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Development of Controller for Te in Force Adjustable Damper

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Abstract- In this report, pole placement controller has been developed for hydraulic semi-active force adjustable damper: 2DOF quarter car model together with 5th order differential equations of damper model is considered. For the purpose of minimizing the car body vertical acceleration in order to improve the ride quality of the car the controller is developed. Four states were investigated. The car body displacement, car body velocity, wheel displacement and wheel velocity. Controllability and stability analysis were performed for the open-loop non-linear and linearized model which lead to the necessity of developing the controller as open-loop system appeared to be unstable. Various control theories on suspension system were investigated in this report, where the challenging part is to improve riding quality while maintaining good handling characteristics subject to different road profile. While developing the pole placement controller, the closed –loop poles were to be placed for both rebound and compression models of the system via MATLAB software. Through simulation, the developed controller proved to be achieving the aims of this project by minimizing the car body vertical acceleration consequently ride quality and comfort can be achieved.

Chapter 1

I. Introduction

a) Review

Vehicle suspension system is one of the most parts in a vehicle. It plays a significant role in physically separating the vehicle's body from the wheels. Hence it supports the vehicle weight, isolate the vehicle body from the wheels. Hence it supports the vehicle weight, isolate the vehicle body from road disturbance and also maintain the traction force between the tire and the road surface (Sam, 2006). A popular and complex problem appears when designing passive vehicle suspension system is the criteria of the system whether it is designed for vehicle handling performance or for passenger ride comfort. When the design of the passive suspension system focuses on increasing the passenger comfort, it's also decreasing the vehicle's abilities to handle road disturbances.

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b) Problem Statement

Passive suspension system is commonly used in passenger's vehicles. The main problem for passive suspension system is the inability give comfort to the passengers without sacrificing the traction force between the tire and the road. Figure (1) describes the relationship between ride quality and vehicle stability or handling performance in a vehicle's passive suspension system. In a addition to that, the performance of a passive suspension system is variable subject to road profile and added passengers weight. That is because the damping force of a passive suspension system is fixed and not adjustable as it has a fixed spring constant and damping coefficient. In this project, hydraulic semi-active suspension system is used with an adjustable damping force.

The main focus is to make the vehicle passenger feel more comfortable without sacrificing the vehicle handling abilities.

c) Objectives of the project

- To develop the mathematical model of a quarter car model
- To develop a controller for a TEIN force adjustable damper
- To evaluate the performance of the controller

d) Methodology

Methodology in conducting project contains a number of steps that must be followed properly in order to achieve project's objectives.

At the beginning, information is to be collected about the project either by literature review or supervision and/co-supervisor's about researching.

Mathematical model is then developed using dynamic motion equation is two mass system with a spring and an adjustable force damper.

After that, MATLAB software is implemented in the system to analyze system stability and to design the controller as well.

Then controller will b collected to the system which will allow both the system and controller to communicate with each other as overall system integration.

Finally, testing analysis in performed where of both precision and accuracy is required in order to

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analyze the performance of the controller to control the system.

e) Report Outline

This report is divided into four chapters. Chapter one is about the background, problem statement, objectives and methodology steps of this project. Chapter two presents the literature review and what other researchers have done in related areas. Chapter three is focuses on the methodology to achieve the project's objectives in details. Finally chapter four is conclusion and recommendation where hardware materials are presented for future work.

f) Gantt chart

A: Registration and title selection.
B: Literature review and research.
C: Mathematical model.
D: Simulation and studies.
E:controller and design.
F:Submission.

Chapter 2

II. Literature Review

a) Introduction

This chapter presents literature review on what other researchers have done in areas are related to this project.

b) What is Modeling and Simulation?

Modeling is developing a level of understanding of the interaction between the system as a whole and some parts in it (Bellinger, 2004). A promote an understanding of the model intends to real system by simplified representation of a system at some particular point in time or space , where simulation is the of a model in such a way that it operates under certain control commands to compress it, thus enabling one to perceive the interactions that would not otherwise be apparent.

c) Vehicle Suspension System

Such a system that contains springs, shock absorber and linkages that connect the wheel to the vehicle body and allows relative motion between the two is called a suspension system (Jazar, 2008). There are three categories of suspension system which are active, semi-active and passive suspension systems. This criteria deepen upon the external power input and/or the control band width into the system.

i. *Passive suspension system*

For a passive suspension system, the damping criteria (spring constant and damping coefficient) are fixed since it consists of non-controlled spring and shock-absorbing damper. Those two elements which are the energy dissipating element (damper) with a fixed –size orifice, and the energy- storing element (spring stiffness) cannot supply energy to the system 9such as fixed –size orifice generates a damping force that is only dependent on the relative velocity of the suspension system), therefore it cannot provide enough control force to improve the handling performance and ride quality for different road disturbances.

To achieve the desired ride characteristics, passive suspension system limits the relative velocity of the sprung mass (car body) and the un sprung mass (wheel) to control the motion. This is done by placing some type of damping element between the body and the wheels of the vehicle, such as hydraulic shock absorber. Properties of the conventional shock absorber establish the trade off between isolating the car body and from the road disturbances to achieve ride quality which is achieved with a soft damping (larger suspension deflection) and at the same time maintaining good wheel-road contact by using hard damping (not allowing unnecessary suspension defection). These parameters are coupled and conflicting. That is, by using a soft damping , it will limit the body acceleration for a comfortable ride , but this reduces the handling performance by allowing more variation in the tire- road contact force. On the other hand, to improve handling it is desirable to minimize the relative velocity by designing a stiffer or higher rate shock absorber.

This stiffness increases the body acceleration but at the same time it decreases the ride quality performance, detract what is considered being good characteristics(Sam, 2006).

ii. *Active suspension system*

The main difference between passive and active suspension system is that active suspension has the ability to supply energy to the system as well as store and dissipate it. Crolla (1988) has categorized the active suspension into two; the low bandwidth (soft) active suspension and the high bandwidth (stiff) active suspension (Sam, 2006). In low bandwidth active suspensions, an actuator is placed in series with the damper and the spring; thus, Wheel hop motion is controlled passively by the damper. Therefore, ride quality is only improved in low-bandwidth active suspension whereas in high-bandwidth active suspension, the actuator is places in parallel with the damper and the spring. High bandwidth active suspension differs from low-bandwidth in controlling both the body motion as well as the wheel hop since the actuator links the un sprung mass to the body.

Therefore, it can improve both handling performance and ride quality simultaneously. Most studies on active suspension system utilize the high –bandwidth type.

iii. *Semi- active suspension system*

According to the control input generation mechanism to the active suspension system, it can be further categorized into another two types which whether to be fully active system or a semi active system. In semi-active suspension system a varying damping force is used as a control force. For example, an electro-rheological (ER) damper or a magneto-rheological (MR) damper applies various level of electrical or magnetic field that causes the various viscosities of the ER or MR fluid. In a hydraulic continuous-damping control (CDC), which is modeled in y thesis, the damper varies the size of an orifice in the hydraulic flow valve driven by a solenoid or stepper motor that is to generate the desired damping forces. To compare between a semi- active and fully active suspension, the weight and cost of a fully active suspension is an obstacle in medium size car since it produces the control force with a separate hydraulic-pneumatic unit. A semi-active suspension is simple, consumes less energy than an active suspension an active suspension and provides better vibration isolation capability than a passive suspension at the body mass resonant frequency.

Two types of dampers are used in a semi-active damper which are the two state dampers and the continuous variable dampers. In the two state dampers, there is a rapid switching in state under a closed-loop control. As for the damper to perform its function of damping the body motion. It is necessary to apply a force that is proportional to the sprung mass velocity. The damper is switched to high state if the body velocity is in the same direction as the damper velocity and switch to low state is the two velocities are in the opposite direction as in this state, the damper is transmitting the input force rather than dissipating energy. The disadvantage of this system is that the rapid switching between states when there are high-velocities generates high –frequency harmonics which lead to undesirable noise (Hasan, 2010).

The characteristic of the continuous variable dampers can be rapidly changed over a wide range. If both damper and body velocity in the same direction, the damper force is controlled to emulate the skyhook damper while if the two velocities are opposite to each other, the damper is switched to its lower rate, which is the closest it will get to the ideal skyhook force. This system has a disadvantage of the difficulty to find devices that have the capability to generate high force at low velocity and low force at high velocity and the rapid changing between those two states. (Hasan, 20d10)

d) Magneto-Rheological (MR) Damper

MR dampers are classified as semi- active dampers. MR fluids are widely spreading in industrial applications such as car suspension, seat suspension, washing machine vibration control and bridge vibration control. MR dampers have been recognized as having a number of attractive characteristics for use in vibration control applications. The fluid properties of MR dampers are not sensitive to contaminates which make MR dampers relatively not expensive to manufacture. In addition to that, MR dampers are reliable, stable and required less power (20-50 Watts) to operate. Because by varying the magnetic field strength can adjust the damping force, so there is no need for mechanical valves which makes the device high reliable. Additionally, the fluid itself responds in milliseconds, allowing the development of devices with a high bandwidth (Laura M.Jansen, 2000).

e) Vehicle Suspension System Control Strategies

This subchapter investigates the various control strategies that have been proposed by numerous researchers to improve the trade-off between ride comfort and handling performance. The search included the areas of semi-active control, MR fluid devices and semi-active dampers. In the following, some of these control approaches will be presented.

i. *Skyhook control*

The skyhook control strategy was introduced by karnopp in 1974 (D.C. Karnop, 1974), which considered as one of the most effective control theory in terms of the simplicity of the control algorithm. Basically, the idea of the skyhook control is linking the sprung mass (vehicle body mass) to the stationary sky by controllable 'skyhook' damper, which can reduce the vertical vibrations by road disturbance as well as generate the controllable force (fskyhook). In their original work, it uses only one inertia damper between the inertia frame and the body mass. Practically , the skyhook control law was designed get an approximation of the force that can be generated by a damper fixed to an inertial reference as the 'sky'.

The skyhook control strategy plays an important role in significantly reducing the resonant peak of the sprung mass so a good ride quality would be achieved. But, in order to improve ride quality and handling performance of the vehicle as well, both of the resonant peaks of sprung mass and unsprung mass need to be reduced. However, the skyhook damper alone cannot reduce both of them simultaneously (Chen, 2009).

(Ahmadian, 1997) discusses how advantage-eous is using a skyhook damper for secondary suspensions. His study shows that the skyhook damper offers more control at one body at expense of less control on the other body. Furthermore, he also introduces hybrid control which is an alternative semi-active control policy that can provide better control of both bodies.

ii. *Semi-active suspension*

Ahmadian, in his study on 1993 (Ahmadian, 1993) examines the effects of semi-active damping on class 8 trucks. Under various damper configurations, the truck was tested on both city streets and highway. It was proven in his study that a configuration of placing semi-active dampers on the front axle and passive dampers on the rear axle is better than placing semi-active

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dampers on the rear and passive dampers on the front axle. As for the ride quality, the result shows nearly equality between placing semi- active dampers on the front axle and passive dampers on the rear axle and placing semi- active dampers on all axles.

In a paper by Yi Chen, he proposed a skyhook surface sliding mode control to the control a semi-active vehicle suspension system for its ride comfort enhancement, in his simulation of the two degree of freedom dynamic model, the results were showing that there was an enhanced level of ride comfort for the vehicle semi-active suspension system with the skyhook surface sliding mode controller (Chen, 2009).

Chapter 3

III. Methodology

a) *Introduction and Methodology Process*

This chapter discusses in details the modeling and simulation of the non-linear semi active suspension system, linearization of the sytem to obtain the state – space model, analysis of the stability and controllability of the linearized system. Finally, the chapter presents the controller design and simulation.

b) *DOF Quarter car model of the non-linear system*

………………

The quarter car model is of the 5th order, here are the equations of motion describing the dynamics of the system:

……………………..

………………………..

Where:

= car body mass

= car body acceleration

= Coil spring constant

= tire mass

= tire acceleration

= tire spring constant

= road displacement (road profile)

f (v,c) = adjustable damping force of the semi-active damper

The adjustable damping force is a function of the relative velocity of the body and the tire (v=…………….) and the number of clicks (c), it is obtained by curve fitting method of the following graph:

The model of the damper is different in rebound and compression, in case of rebound, the relative velocity of the body and tire is going to be positive; the damper model (5th order) is going to be:

……………………………………………………..

……………………………………………

While in case of compression, the relative velocity of the body and tire is negative; the damper model (5th order) is:

Reference: (the 5th order damper model was given to me by my co-supervisor Dr. Fadly Jashi Darvison Bin Ridhuan Siradj)

c) *The state space representation of the non-linear system*

By letting are going to be the states of the plant, the following table describes the normal parameter values used in simulation; the values are approximations of an average car suspension system.

d) *Linearization of the non-linear model*

In this section, we are trying to find the linear approximation of the state space model at the local stability points (equilibrium points)

e) *Controllability and Stability Analysis of the Linearized Model*

In this section, controllability and stability analysis is performed to check whether the new linearized model is stable and controllable.

i. *Controllability and Stability Analysis of Rebound Model*

An important matter is to check whether the open loop system (without any control 0 is stable. To do that, all we need to do is to get the eigenvalues of matrix A, such that those eigenvalues are the poles of the transfer function. To get he transfer function poles, we can simply type –in the MATLAB command eig (A). Figure (11) shows open-loop eigenvalues.

We notice that the open –loop system is unstable since it consists of two poles on the right –half plane.

A system is controllable if there exists a control input that transfers any state of the system to zero in finite time (Introduction: State-Space Methods for Controller Design). Controllability can be proven if and only if the rank of the controllability matrix equals the number of states (Introduction: State-Space Methods for Controller Design). Figure (12) shows (12) shows the controllability of the system:

ii. *Controllability and Stability analysis of compression model*

We can perform the same analysis for the compression model to check whether the open loop system (without any control) is stable. To do that, all we need to do is to get the eigenvalues of matrix A, such that those eigenvalues are the poles of the transfer function. To get the transfer function. To get the transfer function poles, we can simply type in the MATILAB command eig(A). Figure (13) shows the system eignevalues.

f) *Controller Design Using Pole Placement*

Pole placement control method is the placing of the closed-loop poles of a plant in pre-determined locations in the s-plane (Sontag, 1988). The location of the poles plays significant role in controlling the characteristics of the system response. In order to implement this method, the system must be controllable which has been already proven in previous sub-sections (3.4.1 and 3.4.2).

In the controller design, we are going to assume a zero reference (r=0), as the controller would be a regulating controller which stabilizes the system and minimizes the overshoot and settling time of the car body vertical acceleration (x2). The control input of pole placement control (u=-kx), sour state space equations would be

$$X=(A-BK)x$$

$$Y=(C-DK)x$$

i. *Controller Design Using Pole Placement for rebound*

Suppose the criteria of the controller were settling time (Ts <0.4 sec) and overshoot (P0%<7%)from these two factors, we can obtain two poles by solving the second order transient response equation:

……………………………………

…………………………………….

By solving equation (17) , we can obtain the first two pole placement method

$$P1 = -10 + 12i$$

$$P2 = -10 – 12i$$

As for the other two poles, they can be placed at -57 and -58 with the consideration of two important factors:

1. Not to choose the closed loop poles far away the open loop poles, otherwise it will demand high control control effort.
2. Not to choose the closed loop poles very negative, otherwise the system will be fast reacting (i.e. it will have a small time constant), as a consequence, it will lead to the amplification of noise and large control effort.

Moreover, all of the four poles can be changed depending on what the closed-loop response would be; hence, the selection of poles has a disadvantage of spending time and effort to obtain the obtain the optimal solution which achieves the minimum overshoot of the body vertical acceleration. However, the previous selection is our final one since it achieves our goal, we would see that in the simulation and results section. MATLAB was used to obtain the controller gain K.

ii. *Controller Design Using Pole Placement for compression*

Suppose the criteria of the controller were settling time (Ts <o.2 sec), and overshoot (P0%<9%), from these two factors, we can obtain two poles by solving the second order transient response equation:

As for the other two poles, they can be placed at -57 and -58; the controller gain for compression is obtained using MATLAB.

$$K= [73.6563 \quad 936.2196 \quad -859.9053]$$

g) Simulation and results

In this section, we will present our data and results for non- linear system model in both rebound and compression. In addition to that, results and simulations of the closed –loop linearized model after adding the pole placement control in both rebound and compression.

i. *Simulation of non-linear model*

In figure (15) the block diagram represents equations (3), (5), (6), (7) and (8) which describes the non-linear model in case of rebound.

Figure (16) shows the non-linear model in case of rebound is unstable, as the vertical acceleration which we need to minimize goes to an extreme large value which is multiplied by 10^{17} in (m/s2), which makes it a challenge to develop the controller to stabilize the system and minimizes the car body vertical acceleration.

As in case of compression, figure (17) show the block diagram which describes equations (4), (5), (6), (7) and (8).

We notice from figure (18) that the system is unstable as well in case of compression, it is obvious that the current model need to be controlled, otherwise it would be dangerous to ride in a car consisting of an unstable suspension system where the vertical acceleration goes to around (9 x $10^{7\ g)}$

ii. *Simulation of closed –loop linearized model*

This subsection presents the results and simulation after adding the pole placement controller to the system. By referring to section (3.5.1) and (3.5.2). We can see that the pole placement control gain has been obtained in both rebound and compression cases.

In figure (19), the block diagram shows the building of SIMULINK blocks after adding the pole placement gain K and after linearization of the system as well.

We can see from the above figures (figure (Error! Reference source not found) and figure (21) how adding the pole placement control stabilizes the system, where we can see the car body vertical acceleration has been obviously minimized. Hence, ride quality is achieved. Furthermore, the settling time and overshoot are being significantly minimizes.

h) Summary of chapter three

At the beginning of this chapter 5^{th} order differential equations describing the car suspension system have been modeled, the 5^{th} order model has been linearized to1st order model after that. Stability and controllability tests wetr conducted for the new model. Then a pole placement controller was designed for both rebound and compression force of the semi-active damper, finally, simulation and results were presented at the end of this chapter.

Chapter 4

IV. Conclusion and Recommendation

a) Conclusion

In a conclusion, all different suspension systems have been studied. Advantages of using a semi-active suspension system were clearly stated over using passive or active suspension system. Various control theories were investigated. The model was non-linear which required lengthy calculations to linearize it using equilibrium (operating) points. Pole placement controller has been implemented.

MATLAB software was used to:

- Model the non-linear system using block diagram in SIMULINK
- Stability and controllability analysis using MATLAB command window
- Develop the pole placement controller using command window
- Obtain the scope of closed –loop system using SIMULINK

From the results of MALTLAB simulation, implementation of the pole placement controller indeed improves the performance of the system due to the decrease in settling time and overshoots percentage which will reduce body vertical acceleration and improve ride quality.

During the project there were some problems encountered, modeling of a non-linear system where the damper's model is in the 5th order was tedious which required a lot of time and effort. MATLAB errors were hard to overcome and required weeks of research to find people encountered similar errors. As for the current time, the scope of this project focuses only in designing the controller for the suspension system and evaluates it using MATLAB software. For the second part, the project will continue on the implementation of controlling the force adjustable damper, so that the experiments can be conducted including simulations and codes. An advantage of my project is that TEIN force adjustable damper has been developed and ready to be used in the lab.

b) Bills of Materials

Components	Quantity	Price
TEIN force adjustable damper	1	Provided in the lab
stem motor (KT35FMI-030)	1	provided in the lab
Arduino UNO board	1	RM79
Motor driver (L298N)	1	RM22
Accelerometer	1	RM 366
DC to DC concerter	1	Owned
LIPO Rechargeable Battery 11.1V 2200mAH	1	RM 85
TOTAL		Total = RM 552

c) Components Selection

i. *TEIN force adjustable damper*

Inside a TEIN force adjustable damper in figure (22), there is a piston moving within an oil filled tube as the car suspension moves. As the piston moves either up or down, the oil flows through an orifice in the piston. A big orifice opening leads to low damping and vise versa. The damper has a knob on it that can be connected to a stepper motor with a series of click to adjust soft to hard damping.

ii. *Stepper motor*

The step motor in figure (23) is mounted on the damper's knob to control the damping stiffiness. It can be connected to the controller through a motor driver because the controller might bum if connected directly to a motor.

iii. *Motor driver (L298N)*

In order to control the large amount of current flow of the stepper motor, a high current motor driver is used which is (L298N). The L298 is a high voltage, high current dual full-bridge driver.

iv. *Accelerometer*

A one –dimensional accelerometer in figure (25) can be used to measure vertical acceleration.

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